

# TWO-PHASE MECHANICAL FACE SEAL OPERATION: EXPERIMENTAL AND THEORETICAL OBSERVATIONS

by

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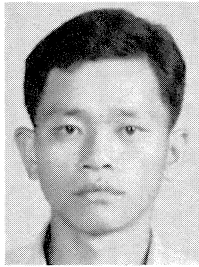
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## ABSTRACT

Experimental and theoretical results related to two-phase mechanical face seal operation are presented. The basic cause for the transition from low friction to high friction operation with increasing temperature is discussed. The causes for seal puffing are discussed. A model which predicts seal performance as a function of temperature is described. Experimental results are presented for a series of high temperature seal tests. These results clearly show the transition with increasing temperature but puffing does not always occur. The experimental results are compared to theory. There is qualitative agreement on the variation of friction torque with temperature and good

agreement on the prediction of the transition point. It is pointed out how the model can be used by seal designers and users to better understand the limits of two-phase operations. Conclusions are drawn and recommendations for further work are made.

## INTRODUCTION

### Background

It has long been recognized that when mechanical seal temperature is above the saturation temperature at discharge pressure, two-phase seal operation results. In some situations this behavior can be beneficial in that it reduces seal friction. In other cases, seal puffing occurs and the seal may be damaged.

It is of considerable practical interest to fully understand two-phase seal operation for the following reasons: 1) many seals operate as two-phase seals by necessity and limits of operation must be known; 2) it may be possible to deliberately operate a seal in the two-phase region to reduce friction and wear; 3) a balanced seal responds to two-phase operation quite differently than an unbalanced seal and this has implications on the balance ratio selected.

### Previous Work

Some progress in developing and understanding two-phase seal operations has been made over the years, both theoretically and experimentally. Some years ago, Lymer [1] presented some experimental results showing when puffing can be expected. He also presented a design procedure which can be used to avoid vaporization of the sealed liquid. Orcutt [2] performed a fairly extensive experimental investigation. He suggested that vaporization of liquid between the faces may be responsible for producing a significantly greater hydrostatic load support and thus reduced friction compared to an all-liquid seal.

More recently, Hughes et al. [3] modeled a seal with a phase change using an idealized heat transfer model. Their results show that the hydrostatic load support in a two-phase seal is greater than for an all-liquid or all-gas seal. There are two points of operation, the stable point being at a higher film thickness than the unstable point. Hughes and Chou [4] have extended this work to the limiting cases of adiabatic and isothermal bounds. They have also included real fluid properties and the effects of radial taper.

Lebeck [5] has also created and presented a comprehensive model for two-phase operations. Load support and friction due to mechanical contact as well as a realistic heat transfer

model are included in the model. The model predicts leakage, friction torque, and relative wear rate as functions of operating temperature. Results for a balanced seal show how leakage becomes large just before the saturation temperature of the sealed fluid is reached. An explanation for puffing is offered.

Most recently, Will [6] has published results of extensive experiments performed on a two-phase seal. His results show clearly a decrease in friction and then an increase as saturation temperature is approached and exceeded.

In this paper, the results from some recent high temperature seal experiments [7, 8] performed at the University of New Mexico are presented and interpreted. The results are compared to theory. Weaknesses in theory and experiment are pointed out. The theory is used to predict performance for some non-parallel face seal cases. Conclusions are drawn and practical implications of these results are discussed.

## THEORY

### General Behavior

To provide some background and a basis for detailed discussion, the basic principles governing two-phase seal operation are reviewed here. Figure 1 shows an outside pressurized mechanical face seal. Figure 2 shows how such a seal might be modeled considering heat transfer and the pressure on the face.

Assuming the seal is operating in the two-phase region, liquid enters between the two faces at  $r_o$ . Pressure drops as the liquid flows radially inward until the pressure corresponds to the saturation pressure at the seal face temperature. At this point,  $r_b$ , the liquid becomes vapor and continues its radially inward flow, exiting at  $r_i$  as a vapor. Now if the liquid consists of more than one component, such as many hydrocarbon streams do, the process is much more complex because there

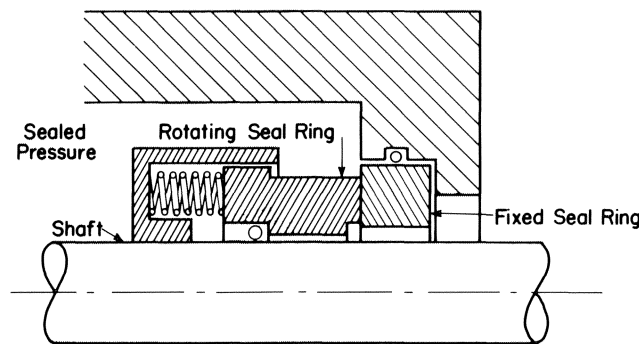


Figure 1. Mechanical Face Seal.

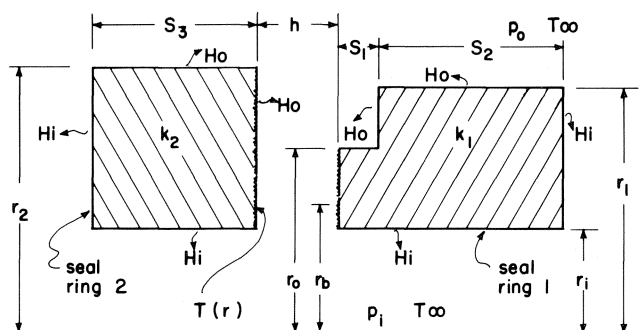


Figure 2. Seal Geometry and Two-Phase Model.

will not be a complete conversion of liquid to vapor at a given radius as in the example cited where water is the liquid.

Figure 3 shows two typical pressure distributions for a seal, one for operation with an all-liquid phase and one with two phases as just discussed. The area under the pressure curve for the two-phase seal is much greater than for the liquid-phase only seal. This suggests that friction might be lower because a greater fraction of the load is supported by fluid pressure as opposed to mechanical contact which produces mechanical friction. The two-phase pressure distribution also shows how the distribution approaches a rectangular distribution (constant pressure across most of the seal face and falling rapidly near the inside radius). It is well known that if the pressure distribution approaches a rectangular distribution in the case of a balanced seal (balance ratio less than 1.0) sufficient load support is developed such that the faces may separate a large amount and excessive leakage will occur. Thus, when a balanced seal is operated as a two-phase seal, particularly near sealed fluid saturation temperature, there is a risk that the seal will open dramatically. A seal with  $B = 1$  or greater cannot open under these circumstances.

### Model

As mentioned, two-phase seal operation has been modeled in some detail [3, 5]. The Lebeck [5] model considers actual geometry and convection heat losses for predicting the seal face temperature. Both mechanical and fluid pressure load support and friction are considered. The stable film thickness at which the seal operates is predicted. Using this model, two-phase operation is predicted to be as shown by Figure 4 for a typical water seal (parallel faces case). Predicted behavior is as described above. Seal friction decreases with increasing temperature until near the saturation point when it suddenly increases. Seal leakage rapidly increases just before saturation conditions. However, the same model applied to other seals shows that leakage does not always rapidly increase but simply decreases as temperature increases toward saturation conditions.

### Puffing

Seals with the rapidly increasing leakage behavior as shown in Figure 4 would be expected to puff as operation passes through saturation. Puffing occurs because the increased leakage cools the seal thus returning its operation to a lower temperature and thus lower leakage. Friction then heats

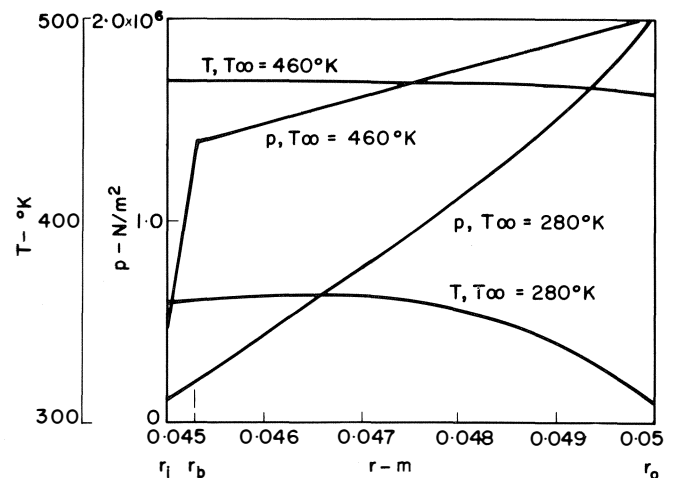


Figure 3. Face Pressure and Temperature Distribution.

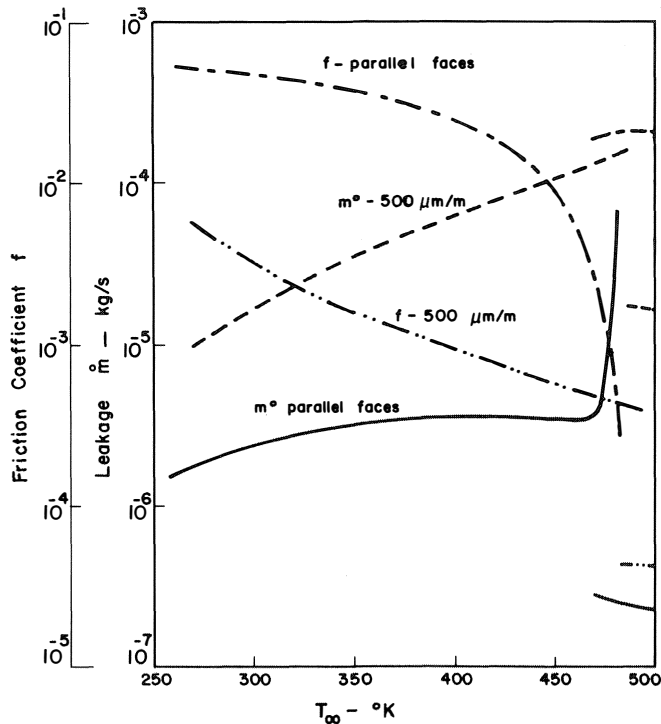


Figure 4. Seal Performance as a Function of Temperature.

the seal again causing it to open and puff. It is thought that this behavior could easily be periodic under certain conditions thus giving rise to what has been observed as puffing. Seals which do not have the increasing leakage characteristic shown would not be expected to puff as operation passed through saturation conditions. An unbalanced seal ( $B \geq 1.0$ ) does not have the increasing leakage characteristics. A balanced seal may or may not, depending apparently on seal geometry.

Figures 5 and 6 show theoretical load support at the face as a function of film thickness at various temperatures. At a given temperature the seal must operate at an intersection of the load support curve and the applied load as shown. The applied load depends on the balance ratio as shown. Only intersections having a negative slope are stable.

Referring to Figure 5 and the 75 percent balance case, as the temperature increases, the right-hand most root moves to the right and eventually disappears. When it disappears, the operation will move to the left-hand most root. The fact that the right-hand root moves out to a large film thickness causes leakage to dramatically increase with increasing temperature and thus leads to puffing as described. At a balance ratio of 0.85 on Figure 5, it would be expected that puffing would not occur because the right-hand root does not move very far out.

In fact, Figure 6 suggests that even at 75 percent balance, puffing may not occur. Figure 6 shows that with increasing temperature the right-hand root simply disappears before the film thickness becomes very large. Detailed results simply show that leakage decreases as saturation is approached. Figure 5 is for a realistic but hypothetical seal. Figure 6 is for the experimental seal discussed later. This theoretical result suggests that relatively small changes in geometry might determine whether or not a seal puffs or more calmly makes the transition through saturation conditions.

#### Radial Taper

All of the theoretical results so far have been for parallel

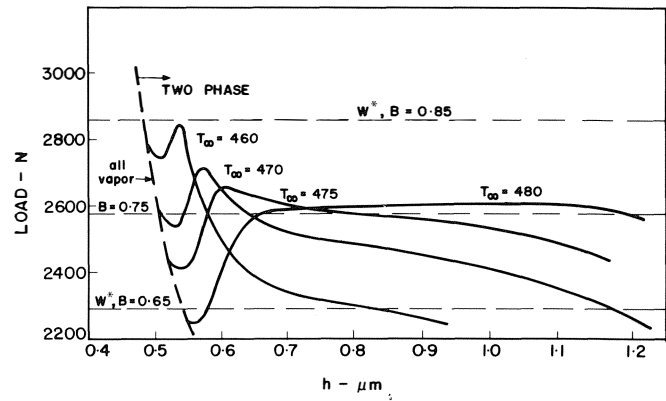


Figure 5. Load Function for Hypothetical Seal.

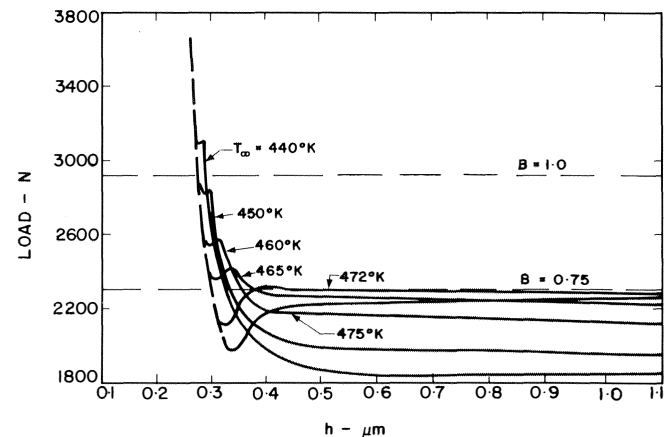


Figure 6. Load Function for Experimental Seal.

face seals. Many times in seal operation, seal faces become non-parallel due to pressure-caused or thermally-caused seal face rotation. Seal instability due to two-phase operation is greatly affected by radial taper. In Figure 4 the performance curves for a convergent radially tapered seal are also shown plotted. Clearly, a puffing type of instability would not occur because there is no sudden increase in leakage. However, leakage is higher throughout the entire range of operation when compared to parallel face operation. This difference in behavior suggests a possible explanation of why apparently identical seals fail due to instability in one application but not in another. Due to pressure or thermal upsets, the net radial taper may simply be different.

## EXPERIMENT

### Apparatus

A series of twenty-nine tests were conducted in water with nominally parallel face seals at elevated temperatures up to 190°C for both  $B = 0.75$  and  $B = 1.00$ . The original purpose of these tests was to provide data to compare to results predicted by the two-phase model [5] just described.

The basic test apparatus is described in detail by [9]. It was originally designed to test wavy seals, but the apparatus can be used more generally. All data acquisition and control are handled by a computer. The seal configuration is shown in Figure 7. A nonrotating, flexibly mounted carbon seal ring is mated with a tungsten carbide rotating ring. Mean seal face diameter is 4 inches.

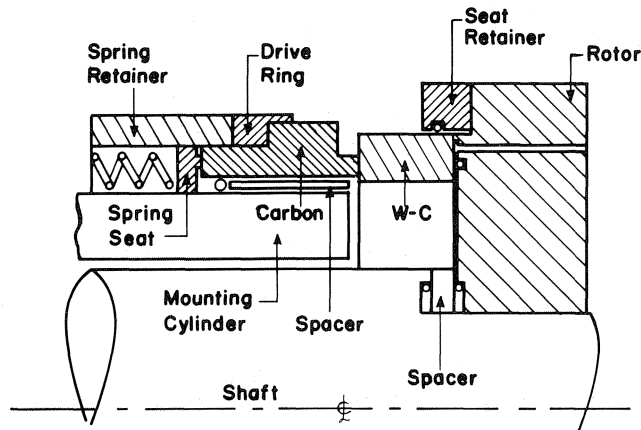


Figure 7. Test Seal Assembly.

Several modifications of the test apparatus were required to make these tests. The first of these was to provide a means to maintain the high temperatures required. After investigating the heat loss through the pressure vessel and the space available inside the pressure vessel, a tubular electric heating element with a capacity of 2 kW was selected and installed inside the pressure vessel. Water temperature is sensed by a thermocouple and monitored by computer. Whenever the water temperature goes above the set temperature, the control program opens a relay which stops the electrical heating.

It was discovered early in this test series that both heating and cooling were necessary to be able to control the temperature under all operating conditions. At the lower test temperatures the cooling system described in [9] was used. The cooling control temperature was set a few degrees above the heating temperature. Experimental results showed that temperature control was in the range of  $\pm 3^\circ\text{C}$  for most of the tests. Torque was measured by a strain gage device as described in [9]. When conducting a test at a high temperature, vapor leakage is expected when two-phase operation is occurring. Thus, a leakage catcher was designed to collect the vapor and to cool it back to the liquid phase. The liquid leakage was collected and measured by a leakage measuring device as described in [9]. The minimum measurable leakage is about  $0.05 \text{ cm}^3/\text{min}$ . Below such rates it is expected that water evaporates before it can be registered. Since no leakage was detected for all of the tests after the seals were worn in, leakage rates below this value must have occurred. A Type J thermocouple was used to measure the seal temperature. The thermocouple was placed about 1 mm beneath the seal face and provides a temperature measurement which is representative of face temperature.

### Procedure

In all of the experiments performed, the outside pressure and the drive speed were adjusted to 1.72 MPa absolute (250 psia) and 1800 rpm, respectively. Ordinarily, the seal was started with the desired pressure and accelerated immediately to the operating speed. The system was heated gradually up to the desired operating temperature within about one hour. Test duration times varied from 1 to 89 hours. Each seal was lapped only once before installation and run through several tests. A computer data acquisition system was used to record and average temperature and torque measurements. Plotted values are based on an average of 12 readings. Such averages were recorded and plotted about once per minute.

### Results

Tests were conducted on four seals with the operating temperature varying from  $37.8^\circ\text{C}$  ( $100^\circ\text{F}$ ) to  $190.6^\circ\text{C}$  ( $375^\circ\text{F}$ ). Test conditions and results are shown in Tables 1 and 2. The data in the tables are averages over the last 8 hours of the test where possible.

Immediately after starting a new or relapped seal, there was a certain period characterized by relatively low torque and seal face temperature during which leakage was detected. This behavior agrees with the theory presented in [8] which suggests that during initial operation, thermal coning produces a convergent film shape and reduces friction. As the face wears flat, friction increases. Within 10 hours or less the mean value of torque and seal face temperature stabilized at a certain level and at this point it was assumed that the seal faces had been worn parallel. No measurable leakage was detected after the run-in period. Torque and seal face temperature were characterized by a wide fluctuation.

Figures 8 through 12 are representative of these tests and the type of data obtained. Figure 8 for test 58 shows the high torque which results for  $B = 1.0$  at  $177^\circ\text{C}$ . Figure 9 shows the results of test 63 where the environment temperature is increased linearly with time for  $B = 1.0$ . It is to be noted how the torque climbs rapidly as the temperature reaches a certain point. Figure 10 is more typical of the test results. Leakage stops after a period of time and torque and temperature are

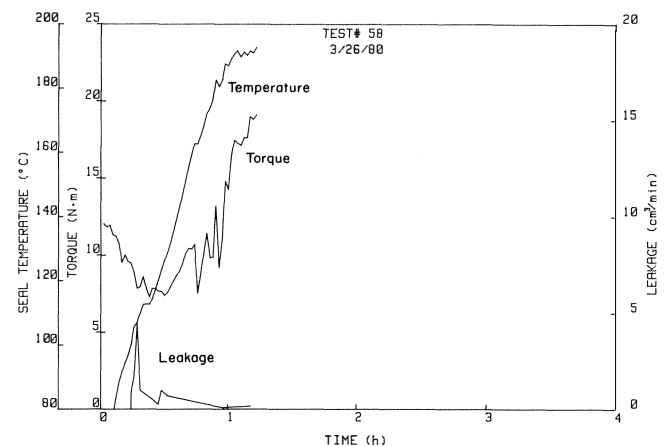
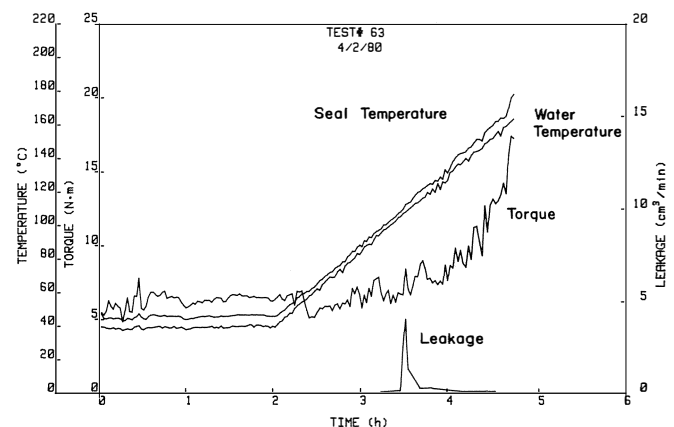
Figure 8. Test 58 –  $T_\infty = 177^\circ\text{C}$ ,  $B = 1.0$ .Figure 9. Test 63 –  $T_\infty$  Variable,  $B = 1.0$ .

TABLE 1. HIGH TEMPERATURE TEST EXPERIMENTAL RESULTS,  $B = 1.00$ ,  $P_{H_2O} = 1.72$  MPA, 1800 RPM.

Test No.	Seal No.	Test Duration (h)	$T_{\infty}$ (°C)	Average Torque (N-m)	Average Face Temp (°C)	Average Leakage (cm <sup>3</sup> /min)
46	5	29.7	93.3	6.8	98.5	0.0
47	5	24.9	121.1	9.1	125.2	0.0
48	5	29.4	148.9	8.9	154.1	0.0
49	5	32.7	121.1	8.2	125.8	0.0
50	5	15.3	148.9	8.6	152.8	0.0
51	5	23.9	93.3	12.8	100.5	0.0
52	5	41.2	65.6	11.1	73.5	0.0
53	5	46.5	37.8	11.4	45.7	0.0
54	7	72.0	93.3	9.1	98.7	0.0
55	7	26.8	137.8	6.8	140.7	0.0
56	7	43.7	165.6	7.4	168.9	0.0
58	9	1.2	176.7	20.9	192.2	0.5
59	9	0.8	176.7	21.1	192.5	0.0
60	9	44.2	165.6	8.3	171.0	0.0
61	9	46.0	137.8	10.9	145.7	0.0
62	9	4.8	37.8 to 176.7			
63	9	4.7	37.8 to 176.7			

TABLE 2. HIGH TEMPERATURE TEST EXPERIMENTAL RESULTS,  $B = 0.75$ ,  $P_{H_2O} = 1.72$  MPA, 1800 RPM

Test No.	Seal No.	Test Duration (h)	$T_{\infty}$ (°C)	Average Torque (N-m)	Average Face Temp (°C)	Average Leakage (cm <sup>3</sup> /min)
69	10	24.1	37.8	4.6	44.1	0.2
70	10	24.1	93.3	6.9	98.1	0.0
71	10	24.0	148.9	6.7	153.2	0.0
72	10	89.2	37.8	1.7	41.1	0.0
73	10	5.0	37.8 to 190.6			
78	10*	37.1	37.8	7.6	47.1	0.0
79	10	12.2	65.6	5.3	71.0	0.0
80	10	10.0	93.3	7.4	100.8	0.0
81	10	12.0	121.1	6.5	126.7	0.0
82	10	11.7	176.7	4.2	180.4	0.0
83	10	3.8	148.9	4.0	152.8	0.1
84	10	2.1	190.6	10.1	199.7	0.0

\*Lapped flat after Test 73.

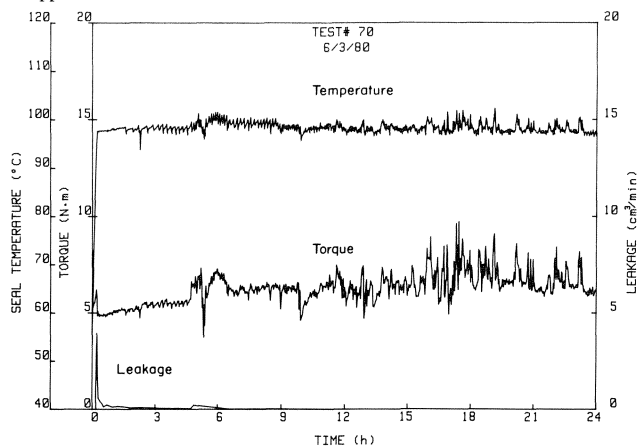


Figure 10. Test 70 –  $T_{\infty} = 93^{\circ}\text{C}$ ,  $B = 0.75$ .

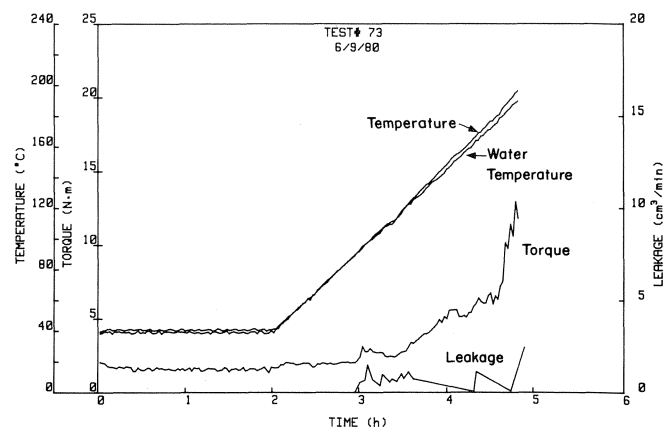


Figure 11. Test 73 –  $T_{\infty}$  Variable,  $B = 0.75$ .

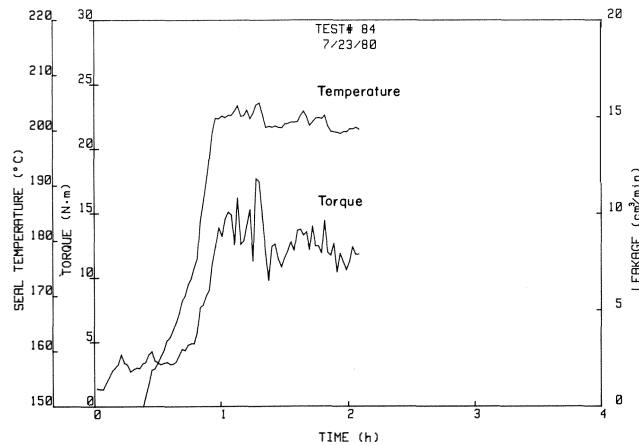


Figure 12. Test 84 -  $T_{\infty} = 191^{\circ}\text{C}$ ,  $B = 0.75$ .

characterized by wide fluctuation. Figure 11 shows the results for test 73 where again the environment temperature is increased linearly with time for  $B = 0.75$ . Figure 12 for test 84 shows the very high torque which results at  $191^{\circ}\text{C}$  for a  $B = 0.75$  seal.

Tests 58 and 59 are of short duration because of a shut down due to overload of the drive motor immediately after the operating temperature reached  $177^{\circ}\text{C}$  ( $350^{\circ}\text{F}$ ). Test 73 was shut down manually when puffing occurred at  $T_{\infty} = 191^{\circ}\text{C}$ . Test 84 was run at a steady  $191^{\circ}\text{C}$ , but no puffing occurred. This is in contrast to test 73 which ended at the same temperature.

Figures 13 and 14 show a different interpretation of the results of tests 62 and 73 where the seal environment temperature was increased linearly with time.  $\Delta T$  is proportional to torque. These tests were an attempt to get a whole range of operating conditions within the same test. Figure 13 shows a small decrease and then a sharp increase with increasing temperature. Figure 14 shows a similar trend but the increase does not occur as quickly.

The torque curves of tests 69, 72, and 79 were quite steady and torque levels were much lower compared to the others. It is thought that these seals were operating with a convergent taper. This caused a sufficient hydrostatic load support so as to prevent wear in. Thus, these tests are excluded from the data which is used later for comparison to theory.

#### Comparison of Experimental Results to Theory

Using the model discussed previously and given by [5],

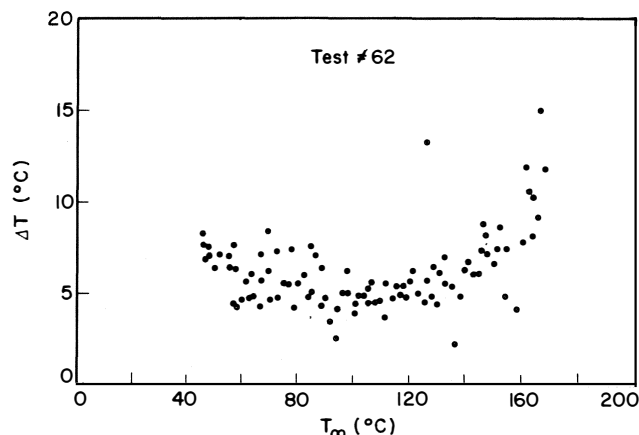


Figure 13. High Temperature Continuous Test,  $B = 1.0$ .

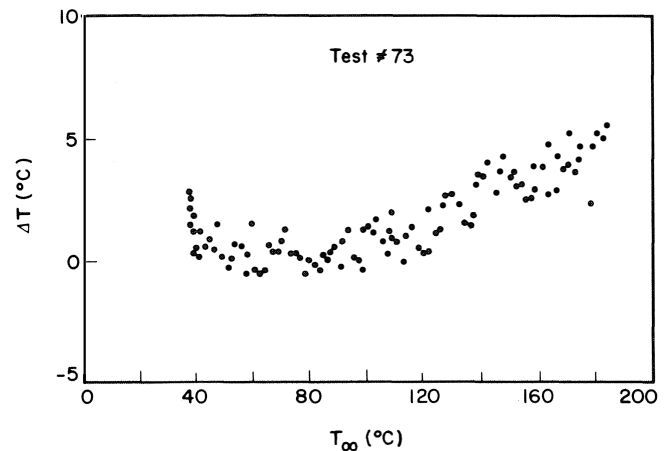


Figure 14. High Temperature Continuous Test,  $B = 0.75$ .

one may calculate seal performance as a function of temperature. Such calculations were made for the experimental seal just described and these calculations are discussed in detail in [7] and [8]. For comparison purposes, a friction coefficient of 0.15 was used in the calculations. Convection coefficients were determined from published data on rotating cylinders. Since no steady state leakage was observed, no computed leakage data is presented.

Figures 15 through 18 show the comparison of theory to experiment as well as a graphical presentation of the experimental data. Figures 15 and 16 show torque as a function of

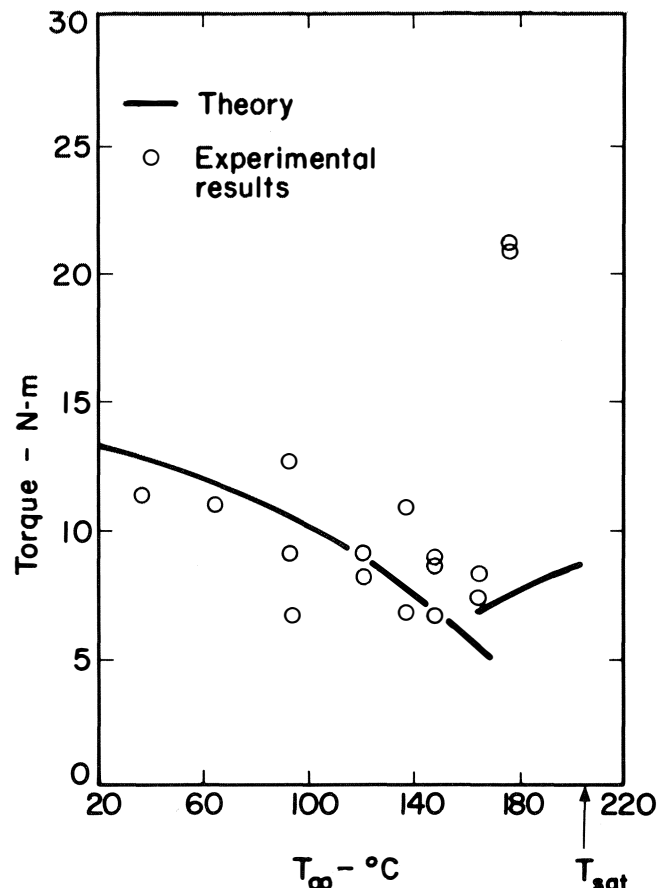


Figure 15. Torque as a Function of Temperature,  $B = 1.0$ .

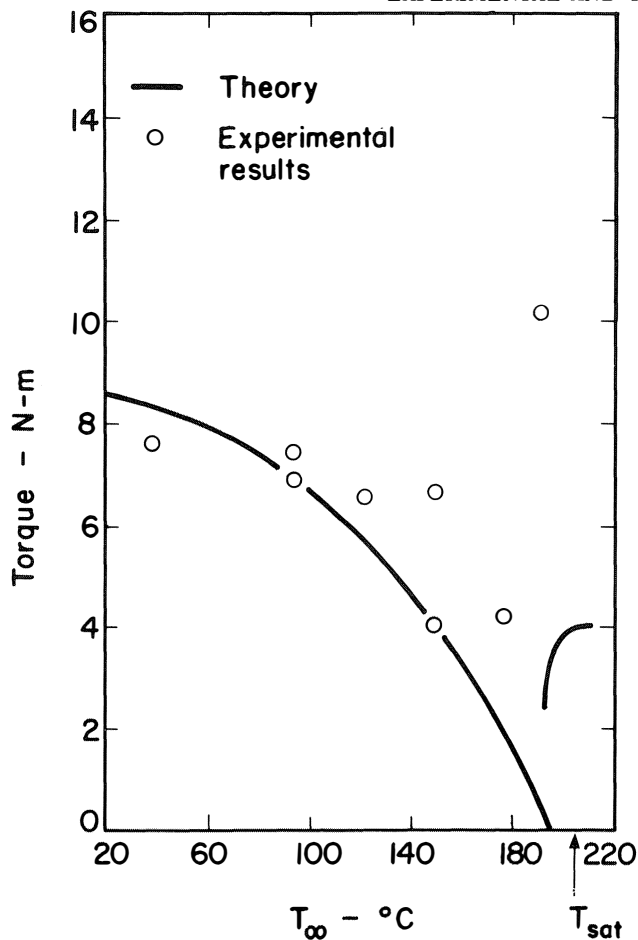


Figure 16. Torque as a Function of Temperature,  $B = 0.75$ .

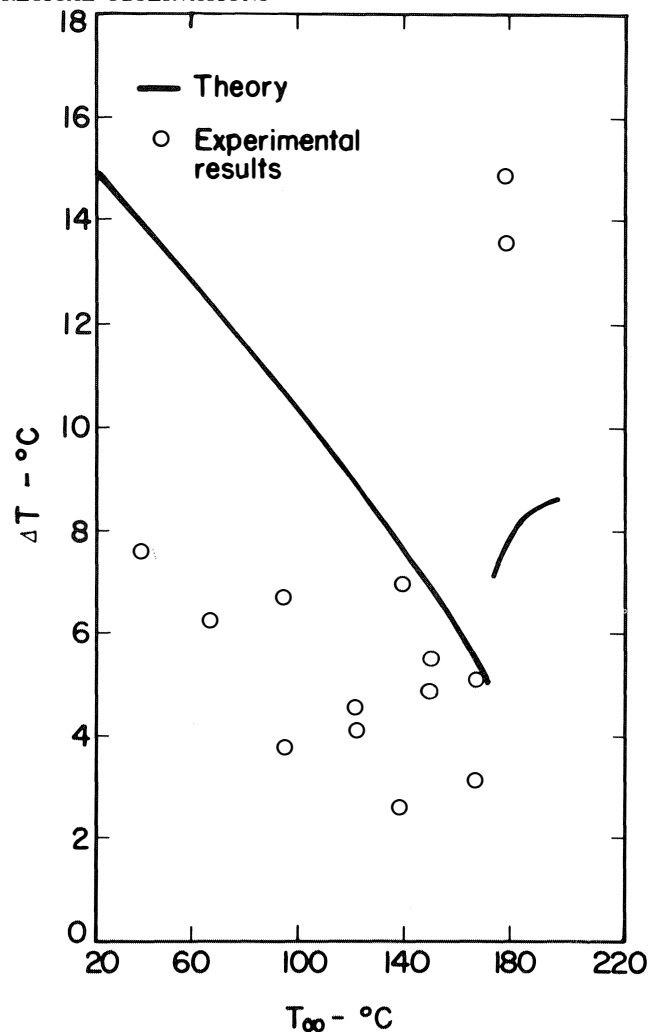


Figure 17.  $\Delta T$  as a Function of Temperature,  $B = 1.0$ .

environment temperature for  $B = 1$  and  $B = 0.75$  respectively. In Figure 15 there is some agreement in that friction torque decreases with increasing operating temperature up to a point. Then torque suddenly increases. Theory greatly underestimates the increase in torque, but it does indicate closely the temperature at which it occurs. The high torque values near  $T_\infty = 180$  are quite repeatable, so the theory is clearly deficient on this point. Figure 16 for  $B = 0.75$  shows similar agreement, but somewhat greater scatter in the experimental results. Again, the transition point is predicted.

Figures 17 and 18 show  $\Delta T$  as a function of temperature.  $\Delta T$  is the measured face temperature minus the environmental temperature and has been shown to be consistently proportional to torque as expected. This measure was used because it was thought that the temperature measurements should be more reliable than the torque measurements, thus the use of temperature measurements should reduce the scatter in the data. Figures 17 and 18 show that the scatter is not reduced at all. In fact, in Figure 18 for  $B = 0.75$ , the data itself does not suggest a curve of the shape predicted by the model. Figure 17 shows a disagreement which can be eliminated by using a lower friction factor in the model. The predicted trend is the same. The point to be made is that the data have been verified by two independent measurements and there is wide scatter in both. In spite of the scatter, the general trend of behavior predicted by the model is supported by the data.

## CONCLUSIONS

Based on the information presented certain conclusions can be drawn.

1) Our basic understanding of how a two-phase seal operates for a simple fluid like water and for parallel faces is satisfactory. That is, seal friction will generally decrease with increasing temperature because of a greater load support caused by the two-phase pressure distribution. The amount of decrease may be small. At some temperature, friction will greatly increase as vapor extends across the entire face. Experiment and theory are in good agreement on this general behavior.

2) The transition above may or may not be accompanied by puffing. For balance ratio 1.0 and greater seals, puffing cannot occur. Both theory and experiment suggest that puffing may or may not occur at lower balance ratios depending upon design. For seals which must operate at or near the transition point, it would seem that the answer to this stability question would be very important for a given seal. Theory suggests the difference between a puffing or destructive type of transition and smooth transition has to do with the shape of the load versus film thickness curve. The shape of this curve is clearly a function of design. These observations suggest that a series of

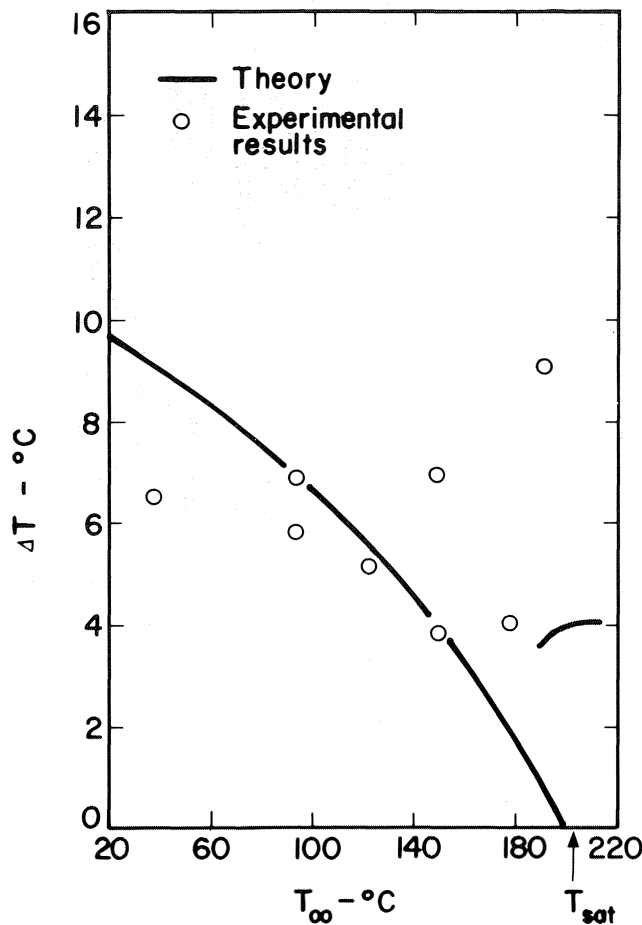


Figure 18.  $\Delta T$  as a Function of Temperature,  $B = 0.75$ .

experiments should be conducted to distinguish between puffing and smooth transition seals. These experiments could be guided by the theory mentioned.

3) The model described can be used to predict the transition temperature for a simple fluid. Experiment verifies the model. This means that one can at least predict the transition temperature for a given seal installation so that the proper amount of cooling can be determined.

4) A converging radial taper greatly alters predicted behavior. The results show how a taper can cause the same seal to undergo a non-puffing type of transition with increasing temperature as opposed to a puffing type or unstable transition. Thus, if a seal has significant thermally-caused rotation or pressure-caused rotation, or if the amount of taper varies because of wear, two-phase performance may be radically

altered. The friction is lower and leakage is higher both before and after the transition. The transition occurs at the same temperature.

#### ACKNOWLEDGMENT

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